Design and Development of Liquid Nitrogen Storage Vessel Using ASME Boiler and Pressure Vessel Code

By

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CERTIFICATE

This is to certify that the thesis entitled, **"Design and Development of Liquid Nitrogen Storage Vessel Using ASME Boiler and Pressure Vessel Code"** submitted by **Mr. Rajendra Kumar Praharaj** in partial fulfillment of the requirements for the award of Master of Technology Degree in **Mechanical Engineering** (specialization of **Machine Design and Analysis)** at National Institute of Technology, Rourkela, Odisha (INDIA) is an authentic work carried out by him under our supervision and guidance. To the best of our knowledge, the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any degree or diploma.

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CONTENTS

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ABSTRACT

A pressure vessel is a closed container which is designed to store liquid or gas at a pressure or temperature, which is different from the ambient temperature and pressure. During operation, the pressure vessel has to withstand several induced stresses due to internal or external pressure. Thus, for the safety purpose storage vessels has to be designed according to ASME standards and rules.

In most of the liquid nitrogen storage vessel, there was no proper control over the heat loss and differential pressure even in static condition which leads to fatal accidents and unnecessary loss of the LN_{2}. Thus, it is important to design the fluid storage container such that no leakage can take place as well as the container has to withstand desired pressure and high or low temperature.

The design mainly concerned with two chambers mounted concentrically out of which one experiences internal pressure and other experiences external pressure with proper fixture and connecting arrangements. The operating pressure is 0.1 MPa for both inside nitrogen storage vessel and outside vacuum jacketed vessel. The present work explores the proper design guidelines for the design of storage vessel which can which can withstand the differential pressure with minimum heat loss using ASME codes and standards. ASME Boiler & Pressure vessel code (ASME Sec VIII, Div-1, Edition 2010, Addenda 2011) has been used for the design of the vessel and materials are selected as per ASME Sec II Part A & D (M), Edition 2010. The connecting pipes are designed as per the ASME B 36.10. Finite Element Analysis and experimental testing have been carried out for validation.

Keywords: ASME Code; Pressure vessel; Finite Element Analysis; Cold Shock Test; Factor of safety; Von mises stress, Hydrostatic test, Cold Shock test,

LIST OF FIGURES

LIST OF TABLES

CHAPTER 1

INTRODUCTION

A liquid nitrogen storage vessel is a close container like pressure vessel which is designed to store or transit fluids at a temperature and pressure which is different from ambient temperature and pressure. Cryogen reservoir like helium, nitrogen storage container are usually have a complex geometrical structure having discontinuities and are required to work under complex loading systems like internal pressure, external pressure, thermal loads, etc. Differential pressure is dangerous and which causes so many accidents in the history of pressure vessel developments. The design of pressure vessel does not mean the detail calculation for defining the dimensions of a member rather to know the reasoning behind the different modes of the pressure vessel failure or the method of stress analysis which gives the significance results and the selection of material type and its environmental behavior.

So the design of the pressure vessel involves some design parameters such as maximum safe operating pressure, corrosion allowance, minimum design material temperature, nondestructive testing like radiography, ultrasonic testing, and pressure testing like hydrostatic test, etc. Thus design, manufacturing, fabrication and testing of these products should follow some national standards, international codes that provide high safety factor and taking care of these above design parameters. Some popular codes which are used for the design of pressure vessel are ASME Boiler and Pressure vessel Code [1], BS 5500, EN13445 etc.

Generally, pressure vessels are in the system of cylinders, spheres, ellipsoids or a combination of these. Liquid nitrogen storage vessels are composed of a complete nitrogen containing chamber with flange rings covered with vacuum jacketed evacuated chamber, fasteners are used for connecting mating parts. When thickness is insignificant in contrast with a mean diameter ($R_m/t > 10$), vessels are designated as membrane and the stresses are developed due to the loading are called as membrane stresses which are maybe tensile or compressive in nature and these stresses are supposed to be constant across the vessel wall [2]. The membrane or wall of the pressure vessel is supposed to have no confrontation to bending. When the wall of the vessel, offers resistance to bending, bending stress also developed in addition to the membrane stresses, thus in an intricate shape vessel loaded with internal pressure, the membrane stress does not give the satisfactory idea about the true stress- strain condition. Other factors like the effect of supports, types of end covers used for closing the vessel, nozzles, thickness variation and external attachments like piping system all cause uneven stress distribution in the pressure vessel. ASME Boiler and Pressure Vessel provide a large factor of safety at the geometrical discontinuity areas like openings, nozzle intersections, change of curvature, thickness variation etc.

The pressure vessel which is exposed to external or internal pressure, stresses are developed in the wall of the vessel. For thick vessel state of stress is triaxial with three principal stresses [2]

S_x = Meridional stress or longitudinal stress

S_y = Latitudinal stress or Circumferential stress

S_r = Radial stress

In addition to these stresses there may be bending stress and torsion stress developed in the wall of the vessel. For the thin cylinders, the radial stress is so small as compared to the other principal stresses, thus radial stress can be neglected so state of stress is biaxial. But in thick vessel, radial stress cannot be ignored, so state of stress is triaxial. Since ASME Code, Section VIII, Division 1, is using a higher factor of safety is used in the design of the vessel to allow for the “unidentified” stresses.

Frequent pressure vessels and boilers busted in the late 1800s and early 1900s which consequential in the development of ASME Boiler and Pressure vessel code. In 1880, The American Society of Mechanical Engineers was structured to support in the formulation of standard specifications for pressure vessel and steam boilers. In the 1914 first edition of code was published that is ASME Rules for the construction of static boilers. ASME section III [1] has specified the stress limit on the bending and membrane stress. Hechmer and Hollinger [3, 4] has studied the stress behavior at the nozzle - vessel intersection region and also they recommend that used 3-D finite element models to study the stress behavior of a nozzle-to-cylinder intersection structure, the stress distribution in a horizontal pressure vessel using Finite Element Analysis has been studied by S.M Khan [5]

1.1 Objective of the study

I. The optimal design of a pressure vessel which can withstand the specified pressure in accordance with ASME Boiler and Pressure Vessel Code.

II. Vacuum chamber should achieve a high order evacuated vacuum environment of for increasing the mean free path of the molecules.

III. Proper thermal design to minimize the heat loss due to evaporation of liquid nitrogen.

IV. Complete assembly should withstand the hydrostatic test, leak test in vacuum condition and cold shock test.

1.2 Principal components of nitrogen storage assembly

I. Nitrogen vessel assembly

II. Vacuum vessel assembly

III. G10 insulation separator

IV. Multilayer insulation sheets

V. Support structure assembly

VI. Lifting lug

CHAPTER 2

2. Literature review

The broad objective of this chapter is to provide background information relating to the pressure vessel study of the literature. . H.Mayer, H.L Stark and S. Ambrose [6] has studied the effect of parameters like stress intensity range and principal stress range on the fatigue life of the pressure vessel. S.V.Dubal and V.G Patil [7] has designed the horizontal pressure vessel supported on the saddle according to the guidelines given by ASME section VIII, Div 1 and Div 2. P. Petrovic [8] has studied the stress distribution in a cylindrical pressure vessel in which load applied at the free end of the nozzle using Finite Element Analysis. Impact of welding residual stress on the failure of the pressure was studied by M. Jeyakumar [9]. A comparative study between design of pressure vessel by analysis and by formula done by A.Th. Diamantoudis and Th. Kermanidis for high strength steel pressure vessels, [10] The identified areas of the survey include:

- Theory behind the pressure vessel design
- Approaches for design of pressure vessel as per ASME code.
- Factors causing pressure vessel failure

2.1 Failure theories for pressure vessel design

Induced stresses due to loading in any component are meaningless if these stresses are not compared with proper failure theories. Thus the failure theories compare the combined stress due to complex loading pattern with the maximum stress at the elastic limit of the simple tensile test. Generally for the pressure vessel design two important failure theories are used, these are Maximum principal stress theory and Maximum shear stress theory. ASME code, section VIII division 1 uses Maximum principal stress failure theory as a basis of design for pressure vessel [1]. H. Mayer, H.L Strark and S. Ambrose [6] has studied the parameters like stress intensity range and principal stress range on the fatigue design of the pressure vessel.

2.1.1 Maximum Principal Stress Theory

According to this theory, breakdown or failure in a component is depends on the magnitude of induced maximum principal stress due to complex loading. As per this theory failure or yielding starts in a component when the induced maximum principal stress equals to the yield stress of the material from the simple tension or compression test at elastic limit.. Maximum principal stress theory is used to predict failure in brittle materials. This theory can be illustrated graphically (Fig. 2.1.1) for biaxial stress system. In graphical representation uniaxial tensile stress at elastic limit or compressive stress at elastic limit lies on the two coordinates. The thick line region represents elastic range of the component and the region cover with dashed line is the safety factor region according to the ASME code.

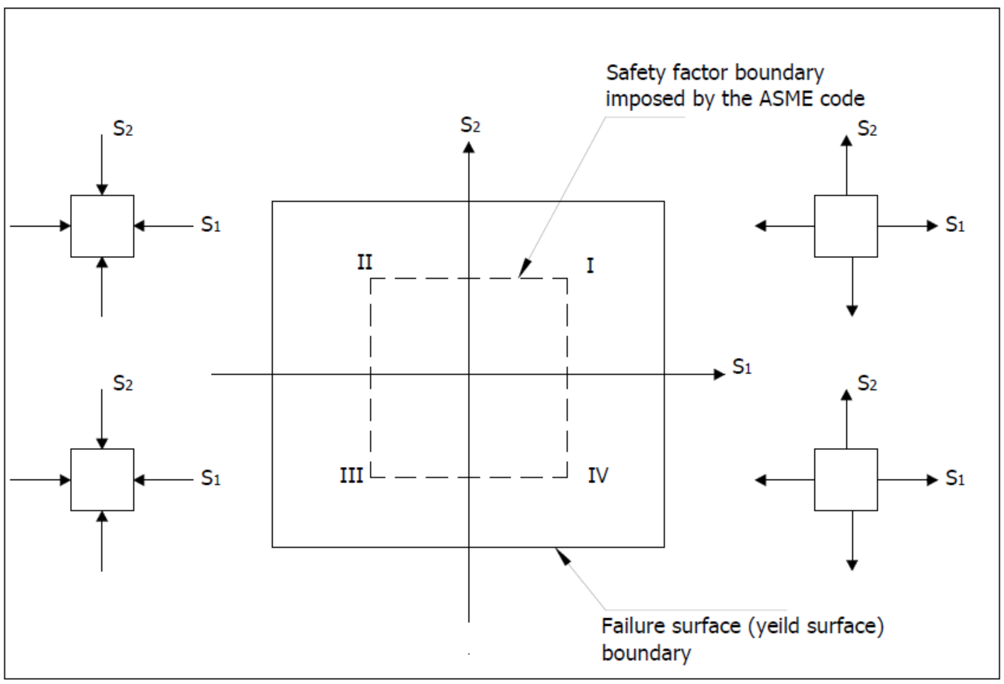


Fig. 2.1 Maximum principal stress failure theory [3]

2.1.2 Maximum Shear Stress Theory

According to this theory, failure or breakdown in a component material is depends on the intensity of the maximum shear stress induced due to complex loading. Thus, the failure starts at a point when maximum shear stress at a point equals to the one half of the yield strength (F_y) from the uni-axial tensile test. Both ASME section III and section VIII, Division 2 use the maximum shear stress criterion. Also this theory closely approaches to experimental results. For the biaxial stress system where $S_1 > S_2$, the maximum shear stress will be $(S_1-S_2)/2$, thus failure occurs when $(S1 – S2)/2 = F_y/2$.

In triaxial stress system, where S1 > S2 > S3, the maximum shear stress will be $(S_1-S_3)/2$, thus failure occurs in the component when $(S1- S3) / 2 = F_y/2$. Generally this theory is used to predict failure in ductile materials.

This theory can be illustrated graphically (Fig. 2.1.1) for the four states of biaxial stress system. In graphical representation uniaxial tensile stress at elastic limit or compressive stress at elastic limit lies on the two coordinates. The outer boundary represents elastic limit stress of the material.

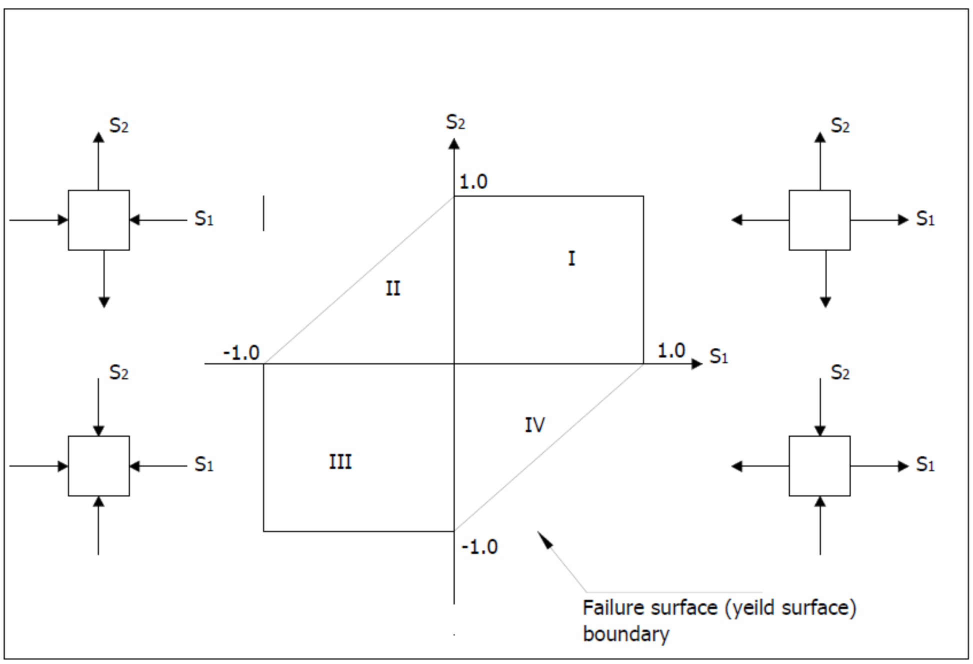


Fig. 2.2 Maximum shear stress failure theory [3]

From the above two theories, the deviation in results between these two theories is more when the magnitude of both the principal stresses is equal. So the thickness formula in ASME Code, section

VIII, Division 1 makes a small difference whether the maximum shear stress theory or maximum principal stress theory is used. Thus, for the thin walled pressure vessels, these two theories give approximately the same results because in thin walled pressure vessel the radial stress is negligible as compare to the other principal stress and that can be ignored and a biaxial stress system is assumed to exist.

ASME Code, Section VIII, Division 2 and ASME Code, Section II use the stress intensity concept which is defined as the doubled the maximum shear stress.

2.2 Broad outlines in ASME code

The complete organization of ASME code has eleven sections which are as follows:

Section I: Has rules for construction of power boiler likes electric boiler, miniature boilers, water boiler for high temperature application and power boiler for locomotive etc.

Section II: It has four parts for material specification and selection as follows:

Part A: Specification for ferrous material

Part B: Specification for nonferrous material

Part C: Specification for filler metals, electrodes, and welding rods.

Part D: Specification for material properties like mechanical, chemical etc.

Section III: Has rules and guidelines for the components used in nuclear industries.

Section IV: Has guideline for design, fabrication and testing of steam boilers.

Section V: Has rules for nondestructive examination to detect weld defects, material defects etc.

Section VI: Has guidelines for the proper care during the operation of the heating boilers.

Section VII: Rules for the proper care during the operation of the power boilers.

Section VIII: It has three divisions according to the operating pressure ranges which has rules and guidelines for design, fabrication, testing, and inspection of pressure vessel.

Section IX: Has rules for welding and brazing qualification also welder and brazing operator qualification

Section X: Has rules for the design of reinforced plastic pressure vessel

Section XI: Provide guidelines for inspection during the operation of nuclear power plant components.

2.3 Factors causing pressure vessel failure

The following factors are causing the premature failure of the pressure vessel so we have to take care these factors in the design stage of the liquid nitrogen storage container.

Material:

- Unsuitable selection of materials or faults in material

Design:

- Due to incorrect or insufficient design data
- Inaccurate design methods
- Inadequate testing facilities

Manufacturing Processes:

- Improper fabrication procedure, including welding
- Poor quality control
- Improper heat treatment process
- Improper forming methods

Service:

- Changes in service condition during its service period
- Inexperienced or unskilled operators or maintenance personnel

Types of Failures [3]

- Failure due to elastic deformation

- Failure due to plastic deformation
- Failure due to brittle Fracture
- Failure due to creep or fatigue
- Failure due to stress corrosion
- Failure due to corrosion fatigue

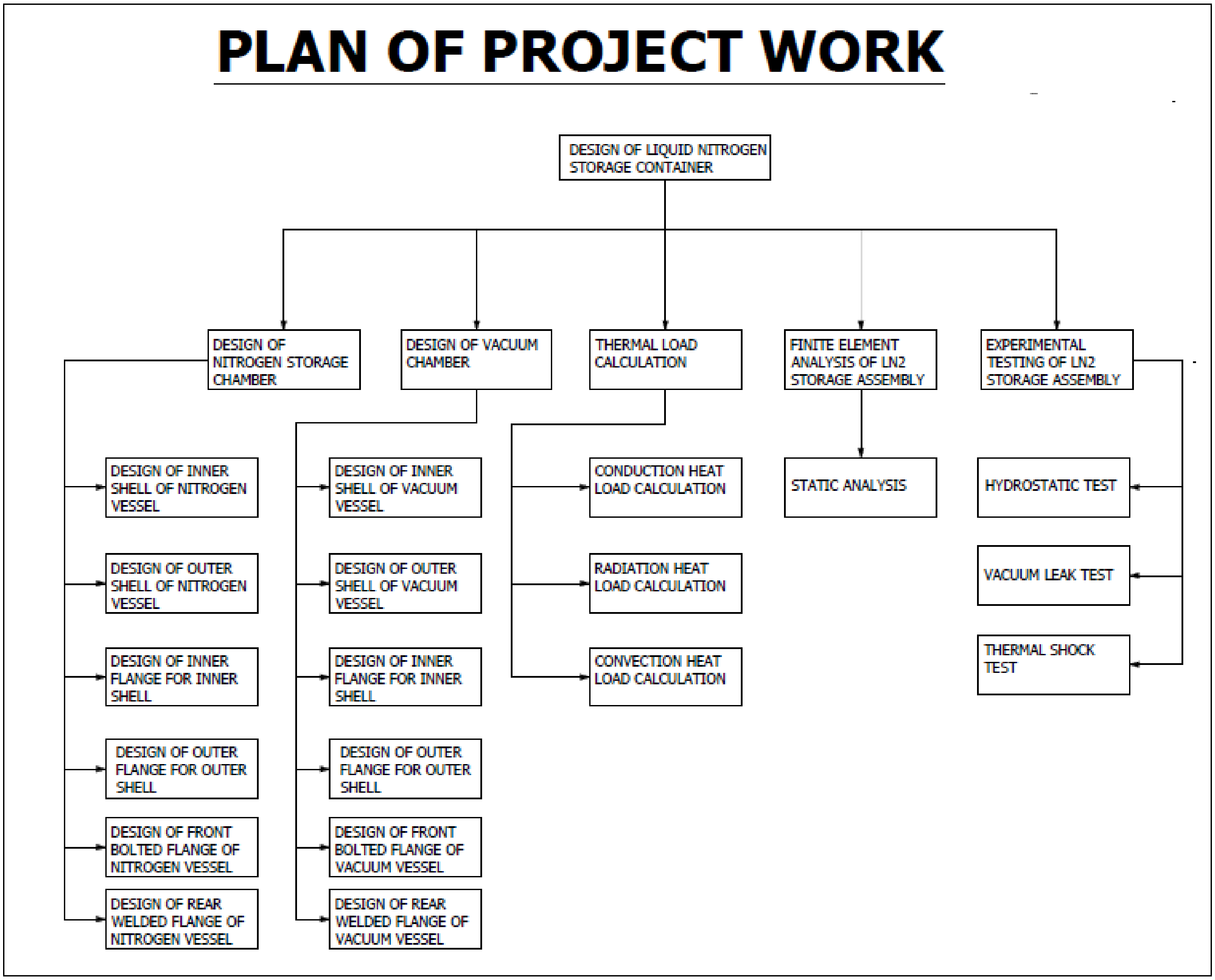
2.4 Concept behind the development of liquid nitrogen container

The boiler temperature of liquid nitrogen is 77 K. The heat load can be transferred to the liquid nitrogen storage container by three different modes that are heat transfer by the conduction mode, heat transfer by the convection mode and the heat transfer by the radiation mode. The maximum amount of heat transferred to the liquid nitrogen storage container is due to natural convection from atmospheric condition (at 300K), hence to diminish this heat load it is required to evacuate the space between vacuum jacketed vessel and liquid nitrogen storage container. The space is evacuated using pumping action by the roughing pump or turbo pump to create vacuum in the range of 10^{-5} mbar. When vacuum created, the number of gas molecules reduces so the mean free path of the molecules increases at 10^{-4} mbar pressure the mean free path is about 100 cm, which reduces the frequent collision of the gas molecules so heat loss due to gas conduction reduces drastically to few milliwatts as vacuum order increases (> 10^{-5} order). The heat transfer coefficients varies proportional to the pressure inside the vacuum chamber, so at high order vacuum condition less amount gas conduction heat load is coming to the liquid nitrogen container.

It is very difficult to maintain such low temperature so, least amount of heat should transfer through the convection from atmosphere (at 300K) to container (at 77 K) carrying liquid nitrogen thus, generally nitrogen storage vessels are vacuum jacketed or double walled to keep separate the container carrying LN2 from atmospheric conditions. The vacuum space between the nitrogen storage vessel and vacuum jacketed vessel, reduce the heat loss due to conduction and convection.

Also, some multilayer insulation made of aluminum foil having high reflectivity are used to reduce heat loss due to radiation. Some G10 insulation material having a very low thermal conductivity is used between two vessels for reducing heat loss due to conduction.

2.5 Project Work Plan Chart



CHAPTER 3

3. Design of Nitrogen container

In general pressure vessels are designed in accordance with ASME Code. Design of nitrogen container mainly contains design of inner and outer vessel, design of front bolted flange, design of front cover flange and rear cover flange. ASME section VIII division 1 has been used during the design of each component.ASME section II has been used for the selection of material for each component.

Table 3.1 Reference codes & Standards

Sr. No	Code	Description
1	ASME Sec VIII, Div-1, Edition 2010, Addenda 2011	Vacuum chamber & Nitrogen Chamber
2	ASME Sec II Part A & D(M), Edition 2010	Materials
3	ASME B 36.10	Seamless Pipe

Table 3.2 Design data for Nitrogen vessel

Sr. No	Description				
1	Design Code	ASME Sec VIII, Div 1, Edition 2010, Addenda 2011			
2	Working Pressure for Nitrogen vessel, Gauge (Internal)	1	bar	0.1	MPa
4	Design Pressure for Nitrogen vessel	2	bar	0.2	MPa
5	Design Temperature	313	K	40	^{o}C
6	Maximum Allowable Working Pressure	2	bar	0.2	MPa
7	Corrosion Allowance			0	mm
8	MDMT	77	K	-196	^{o}C
9	Radiographic requirement	Full			
10	Joint Efficiency	1			
11	Service	Liquid Nitrogen Chamber			
12	Hydrostatic Test Media	Water			

The Following Unit Conversions Factors are used in this report

1. To convert MPa to Psi, Divide MPa value with 0.0068948 (ASME Sec. VIII Div.1, Table GG-3)

2. To Convert Bar to Psi Divide bar value with 0.06894757 (ASME Sec. VIII Div.1, Table GG-3)

Based on the above conversions, to convert Bar to MPa multiply with 0.10000

1 MPa = 145.0368 Psi

1 Bar = 14.50377 Psi

i.e. 1 Bar = 14.50377 / 145.0368 = 0.10000 MPa

Also 300 K = 27 ^oC; 80 K = -193 ^oC

3.3 Design internal pressures calculation.

Density of Contents, g (Kg/m^3)	=	1000.00 (*Assumed as water density*)
Static Pressure Head	=	g x 9.8067 x H
	=	1000 x 9.8067 x 1615/1000 = 15837.82 Pascal
	=	15837.82 x 10^{-6} Mpa
	=	0.015Mpa
Design Internal Pressure	=	Design Pressure + Pressure due to Static Head
	=	0.2 MPa + 0.015 MPa
	=	0.215 MPa

3.4 Hydrostatic pressure calculation

Hydrostatic Test Pressure = Maximum allowable working pressure x LSR x 1.3

LSR = Lowest Stress Ratio (From B.1)

LSR = Stress value at test Temperature / Stress value at design Temperature = 1

Hydrostatic Test Pressure for Nitrogen vessel = 0.20 x 1 x 1.3

= 0.26 MPa

Hydrotest to be carried out in a horizontal position only**.**

Table 3.5 MATERIAL OF CONSTRUCTION EVALUATION CHART AS PER UG-23, UHA-23, TABLE UHA 23 OF ASME SECVIII DIV-1 AND ASME Sec-II Part D,TABLE 1A

No	Component type	Material of construction	Table No	Allowable Stress (MPa)		
				MDMT (0^{O}C)	Design Temp (40 ^{O}C)	Hydrostatic temp (48^{O}C)
1	Inner & Outer Shell of nitrogen chamber	SA 240 Type 316L	1A	115	115	115
2	Inner & Outer Shell Flanges of nitrogen vessel	SA 240 Type 316L	1A	115	115	115
3	Nozzle	SA 240 Type 316L	1A	115	115	115
4	Nozzle Flanges	SA 240 Type 316L	1A	115	115	115
5	Bolting for nitrogen vessel	SF 468	3	155	155	155

3.6 Design of Outer cylindrical shell (Nitrogen Vessel) Thickness under Internal Pressure [Ref: UG-27 & Appendix-1]

The material selected for the outer cylindrical shell is SA 240 TYPE 316L. Assumed thickness (t) of the vessel is 5 mm. Considering the mill tolerance of 0.19 mm from the UG 16 (c), table A 3.5, SEC II Part A, so effective thickness is 5 – 0.19 = 4.81 mm.

Considering thinning due to rolling = 4%

Reduction in thickness due to rolling = 4.81 - (0.04x4.81) = 4.61

Internal design pressure (P) = 0.21 MPa

Maximum allowable stress from section II part D (S) = 115 MPa

The outer diameter of the vessel (D_0) = 1000 mm

Inside diameter of the vessel (D) = 990 mm

The inside radius of the vessel (R) = 495 mm (taking corrosion allowances is zero)

Longitudinal joint efficiency from UW-12 = 1

The minimum required thickness (t) as per ASME Appindix -1 is given by :

$$t = PR/((SE)-(0.6P))$$

$$= 0.21x500 / (115x1 + 0.6x0.21)$$

$$= 0.896 \text{ mm}$$

But according to ASME Code, UG-16 (b) the minimum thickness of the shell must be 1/16 inch excluding the corrosion allowances i.e 1.5 mm.

Since required thickness is less than the provided thickness, i.e. 1.5 < 4.61 mm, So provided thickness is adequate.

3.7 Inner cylindrical shell (Nitrogen Vessel) Thickness under External Pressure [Ref: UG-28 (c) (1)]

The material selected for the outer cylindrical shell is SA 240 TYPE 316L. Assumed thickness (t) of the vessel is 5 mm. Considering the mill tolerance of 0.19 mm from the table A 3.5, SEC II Part A, so effective thickness is 5 – 0.19 = 7.75 mm.

Outer diameter of the vessel (D_0) = 535 mm

Inside diameter of the vessel (D) = 525 mm

Length of the vessel = 900 mm

The ratio of length of the vessel to the outer diameter of the vessel = L/Do = 900/535 = 1.68

The ratio of the outer diameter of the vessel to the thickness of the vessel = Do/t = 535/8 = 189.5

From the ASME code Fig- g, Section II, Part D, Factor (A) = 0.0007

From the ASME code fig-HA-4, for SA 240 TYPE 316L, Factor (B) = 60

The allowable external pressure working pressure (Pa) from the code is given by:

$$Pa = 4B / (3*(Do/t))$$

= 4 x 60 / (3x(535/4.81))

= 0.72 MPa

The actual working pressure is 0.1 MPa which is lower than the allowable external pressure of 0.72 MPa so the provided thickness (4.81 mm) is adequate.

3.8 Front Bolted Flange (Nitrogen Vessel) Thickness [Ref: UG-34 (2) Unstayed Flat Heads & Covers]

The internal design pressure (P) = 0.2 MPa

The material selected for the outer cylindrical shell is SA 240 TYPE 316L .Maximum allowable stress from section II part D (S) = 115 MPa

Diameter of the opening of the shell (d) = 1005 mm

Required thickness of the shell wall (Tr) = 0.87 mm

Nominal thickness of the shell (Ts) = 5.0 mm

Ratio of Tr/ Ts = 0.87/ 5 = 0.174

Factor (C) depending on the method of attachment = 0.30

Joint efficiency for full Radiography (E) = 1

The minimum thickness (t) required for the flat unstated head [UG-34 (2) eqn-(2)] is given by:

t = d x sqrt (CP/SE)

= 1005 x sqrt (0.3 x 0.2/ 115 x 0.7)

= 24 mm

Thus the provided thickness is 25 mm.

3.9 Rear Welded Flange (Nitrogen Vessel) Thickness [Ref: UG-34 (2) Unstayed Flat Heads & Covers]

The internal design pressure (P) = 0.2 MPa

The material selected for the outer cylindrical shell is SA 240 TYPE 316L. Maximum allowable stress from section II part D (S) = 115 MPa

Diameter of the opening of the shell (d) = 1000 mm

Required thickness of the shell wall (Tr) = 0.87 mm

Nominal thickness of the shell (Ts) = 5.0 mm

Ratio of Tr/ Ts = 0.87/ 5 = 0.17

Factor (C) depending on the method of attachment = 0.20

Joint efficiency for full Radiography (E) = 1

The minimum thickness (t) required for the flat unstayed head [UG-34 (2) eqn-(1)] is given by :

t = d x sqrt (CP/SE)

= 1184 x sqrt (0.3 x 0.1/ 138 x 0.7) = 18.7 mm

Thus the provided thickness is 20 mm.

Table 3.10 Design data table for Inner Flange (Nitrogen Vessel) Thickness [Ref: SEC-VIII DIV-1, MANDATORY APPENDIX-2, CLAUSE: 2-13, LOOSE RING TYPE REVERSE FLANGE]

Type of Flange Selected (Category 3 Flange and Class-1 Assembly)	**Loose Type Flanges Without Hub (Fig-2-13.2)**		
Outside diameter of flange	A	535.0	mm
Inside diameter of shell	B	535.0	mm
Inside diameter of reverse flange	B'	425.0	mm
Bolt circle diameter	C	483.0	mm
Effective thickness of the flange	t	35.0	mm
Diameter of location of gasket load reaction	G	520	mm

Bolt Loads: [Ref: Appendix-2 2-5]

External design pressure	*P*	0.20	MPa	
Total hydrostatic end force	H	42452.8	N	H=0.785G2P
Width of gasket	N	4.0	mm	Solid flat metal : Soft Copper (From Table 2-5.1)
Effective gasket seating width	b0	2.0	mm	b0 = N/2 , From Table 2-5.2 , Facing sketch 1(a)

Basic gasket seating width	B	2.0	mm	b = b0 when b0 < 6 mm
Gasket factor	M	4.75	--	For Solid flat metal : Soft Copper (From Table 2-5.1)
Gasket or joint contact surface unit seating load	Y	90	MPa	For Solid flat metal : Soft Copper (Table 2-5.1)
Total joint contact surface compression load	Hp	6204.6		Hp = 2b x 3.14GmP
Minimum required bolt load for operating conditions	Wm1	48657.4	N	Wm1 = H+Hp
Minimum required bolt load for gasket seating	Wm2	293904	N	Wm2 = 3.14bGy
Allowable bolt stress at atmospheric temp.	Sa	155.0	MPa	From Section II Part D Table 3
Allowable bolt stress at design temp.	Sb	155.0	MPa	From Section II Part D Table 4
Total crossectional area of bolts at root of thread, required for operating conditions	Am1	313.9	mm2	Am1 = Wm1 / Sb
Total crosssectional area of bolts at root of thread, required for gasket seating	Am2	1896.155	mm2	Am2 = Wm2 / Sa
Total required crossectional area of bolts, taken as greater of Am1 & Am2	Am	1896.2	mm2	
Nominal bolt diameter	A	12	mm	For M12 bolt
Cross-sectional area of bolts using the root dia of thread = 28 x 72.39 = 2026.92	Ab	2026.92	mm2	Provided = 28 nos.
Flange design bolt load, for operating conditions	W	48657.4	N	W = Wm1
Flange design bolt load, for gasket seating	W	304038.3	N	W = (Am +Ab)Sa/2

Flange Moments: [Ref: Appendix-2 2-6]

Hydrostatic end force on area inside of flange	H_D	44937.3	N	$H_D = 0.785B^2P$
Gasket load	H_G	6205	N	$H_G = W-H$
Difference between H and H_D	H_T	-2484.53	N	$H_T = H-H_D$
Radial distance from the bolt circle to the circle on which H_D acts	h_D	-26.0	mm	$h_D = (C-B)/2$ From Table 2-6 Appndx-2
Radial distance from gasket load reaction to the bolt circle	h_G	-18.5	mm	$h_G = (C-G)/2$ From Table 2-6 Appndx-2

Radial distance from the bolt circle to the circle on which HT acts	h_T	-22.25	mm	h_T = 1/2(C-(B+G)/2) From 2-13
Component of moment due to H_D	M_D	-1168370	N-mm	$M_D = H_D h_D$
Component of moment due to H_T	M_T	55280.7	N-mm	$M_T = H_T h_T$
Component of moment due to H_G	M_G	-114785.8	N-mm	$M_G = H_G h_G$
Flange moment (for operating conditions)	Mo	-37217.8	N-mm	Mo = $H_G(h_D-h_G)+H_T(h_T-h_G)$
Flange moment (for gasket seating)	Mo	-5624708.6	N-mm	Mo = Wh_G

Calculations of Flange Stresses: [Ref: Appendix-2 2-7]

Factor K	K	1.2588		K= A/B'
Factor involving K	Y	8.567		
Tangential flange stress (For Gasket Seating)	S_T	92.6	N/mm^2	$S_T = YM_0 / t^2B'$
Tangential flange stress (For Operating Condition)	S_T	0.6	N/mm^2	
Radial Flange stress	S_R	0	MPa	For Loose type flanges without hubs
Longitudinal stress	S_H	0	MPa	For Loose type flanges without hubs
Allowable design stress for material of flange at design temperature	Sf	**115.0**	N/mm^2	

As per Appndix-2-8 (a) (2) Tangential Flange Stress S_T (for gasket seating as well as operating condition) which are not greater than Allowable Design Stress Sf (115 MPa). Therefore the provided thickness of flange is adequate.

Table 3.11 Design data table for Outer Flange (Nitrogen Vessel) Thickness [Ref: SEC-VIII DIV-1, MANDATORY APPENDIX-2, LOOSE RING TYPE FLANGE]

Type of Flange Selected (Category 3 Flange and Class-1 Assembly)	**Loose Type Flanges Without Hub (Fig-2-13.2)**		
Outside diameter of flange	A	1090.0	mm
Inside diameter of flange	B	990.0	mm
Bolt circle diameter	C	1042.0	mm
Effective thickness of the flange excluding o-ring groove	t	**45.0**	mm
Diameter of location of gasket load reaction	G	1005.0	mm

Bolt Loads: [Ref: Appendix-2 2-5]

Internal design pressure	*P*	0.20	N/mm^2	
Total hydrostatic end force	H	158573.9	N	$H=0.785G^2P$
Width of gasket	N	4.0	mm	Solid flat metal : Soft Copper (From Table 2-5.1)
Effective gasket seating width	b_0	2.0	mm	$b_0 = N/2$, From Table 2-5.2 , Facing sketch 1(a)
Basic gasket seating width	B	2.0	mm	$b = b_0$ when $b_0 < 6$ mm
Gasket factor	M	4.75	--	For Solid flat metal : Soft Copper (From Table 2-5.1)
Gasket or joint contact surface unit seating load	Y	90	N/mm^2	For Solid flat metal : Soft Copper (From Table 2-5.1)
Total joint contact surface compression load	Hp	11991.7		Hp = 2b x 3.14GmP
Minimum required bolt load for operating conditions	Wm1	170565.6	N	Wm1 = H+Hp
Minimum required bolt load for gasket seating	Wm2	568026	N	Wm2 = 3.14bGy
Allowable bolt stress at atmospheric temp.	Sa	155.0	N/mm^2	From Section II Part D Table 3
Allowable bolt stress at design temp.	Sb	155.0	N/mm^2	From Section II Part D Table 4
Total crossectional area of bolts at root of thread, required for operating conditions	Am1	1100.4	mm^2	Am1 = Wm1 / Sb
Total crossectional area of bolts at root of thread, required for gasket seating	Am2	3664.68	mm^2	Am2 = Wm2 / Sa
Total required crossectional area of bolts, taken as greater of Am1 & Am2	Am	3664.7	mm^2	
Nominal bolt diameter	A	12	mm	For M12 bolt
Cross-sectional area of bolts using the root dia of thread	Ab	3764.28	mm^2	Provided = 52 nos.
Flange design bolt load, for operating conditions	W	170565.6	N	W = Wm1
Flange design bolt load, for gasket seating	W	575744.7	N	W = (Am +Ab)Sa/2

Flange Moments: [Ref: Appendix-2 2-6]

Hydrostatic end force on area inside of flange	H_D	153875.7	N	$H_D = 0.785B^2P$
Gasket load	H_G	11992	N	H_G = W-H
Difference between H and H_D	H_T	4698.23	N	$H_T = H-H_D$
Radial distance from the bolt circle to the circle on which H_D acts	h_D	26.0	mm	h_D = (C-B)/2 From Table 2-6 Appndx-2
Radial distance from gasket load reaction to the bolt circle	h_G	18.5	mm	h_G = (C-G)/2 From Table 2-6 Appndx-2
Radial distance from the bolt circle to the circle on which HT acts	h_T	22.3	mm	$h_T = (h_D+h_G)/2$ From Table 2-6 Appndx-2
Component of moment due to H_D	M_D	4000768	N-mm	$M_D = H_D h_D$
Component of moment due to H_T	M_T	104535.5	N-mm	$M_T = H_T h_T$
Component of moment due to H_G	M_G	221845.7	N-mm	$M_G = H_G h_G$
Flange moment (for operating conditions)	Mo	4327149.4	N-mm	Mo = $M_D+M_T+M_G$
Flange moment (for gasket seating)	Mo	10651277.0	N-mm	Mo = W(C-G)/2

Calculations of Flange Stresses: [Ref: Appendix-2 2-7]

Factor K	K	1.1010		K= A/B
Factor involving K	Y	20.128		
Tangential flange stress (For Gasket Seating)	S_T	106.9	N/mm^2	$S_T = YM_0 / t^2B$
Tangential flange stress (For Operating Condition)	S_T	43.4	N/mm^2	
Radial Flange stress	S_R	0	MPa	For Loose type flanges without hubs
Longitudinal stress	S_H	0	MPa	For Loose type flanges without hubs
Allowable design stress for material of flange at design temperature	Sf	115.0	N/mm^2	

As per Appndix-2-8 (a) (2) Tangential Flange Stress S_T (for gasket seating as well as operating condition) which are not greater than Allowable Design Stress Sf (115 MPa). Therefore the provided thickness of flange is adequate

3.12 Thickness of Straight Nozzle Wall under Internal Pressure (N1) [Appendix 1-1(a)]

The material selected for the outer cylindrical nozzle is SA 240 TYPE 316L. Assumed thickness (t) of the nozzle is 5.74 mm Considering Mill Under tolerance of 12.5 % (Table-3, Sec-II, Part A), so effective thickness is = 5.74 x 0.875 = 5.02 mm.

Internal design pressure (P) = 0.21 MPa

Maximum allowable stress from section II part D (S) = 115 MPa

Inside diameter of the nozzle (D) = 90.12 mm

The inside radius of the nozzle (R) = 45.06 mm (taking corrosion allowances is zero)

Longitudinal joint efficiency from UW-12 = 1

The minimum required thickness (t) as per ASME Appendix -1 is given by:

$$t = PR/ ((SE)-(0.6P))$$

$$= 0.2 \times 45.06 / (115x1 + 0.6x0.21)$$

$$= 0.07 \text{ mm}$$

But according to ASME Code, UG-16 (b) the minimum thickness of the nozzle must excluding the corrosion allowances is 1.5 mm.

Since required thickness is less than the provided thickness, i.e. 1.5 < 5.02 mm, so provided thickness is adequate.

CHAPTER 4

4. Design of Vacuum container

In general pressure vessels are designed in accordance with ASME Code. Design of nitrogen container mainly contains design of inner and outer vessel, design of front bolted flange, design of front cover flange and rear cover flange. ASME section VIII division 1 has been used during the design of each component.ASME section II has been used for the selection of material for each component.

4.1 Reference codes & Standards

Sr.No	Code	Description
1	ASME Sec VIII, Div-1, Edition 2010, Addenda 2011	Vacuum chamber
2	ASME Sec II Part A & D(M), Edition 2010	Materials
3	ASME B 36.10	Seamless Pipe

4.2 Design data for Vacuum vessel:

Sr.No	Description				
1	Design Code	ASME Sec VIII, Div 1, Edition 2010, Addenda 2011			
2	Working Pressure for Nitrogen vessel, Gauge (External)	1	bar	0.1	MPa
4	Design Pressure for Vacuum vessel, Gauge (Internal)	1	bar	0.1	MPa
5	Design Temperature	313	K	40	^{o}C
6	Maximum Allowable External Working Pressure	1	bar	0.1	MPa
7	Corrosion Allowance			0	mm
8	MDMT	273	K	0	^{o}C
9	Radiographic requirement	None			
10	Joint Efficiency	0.7			
11	Service	Vacuum Chamber			
12	Hydrostatic Test Media	Water			

4.3 Calculation of Design Internal Pressure

Static Head Calculation

Design Pressure, Gauge (MPa) = 0.1

Static Head, H (mm) = 1317 mm

Density of Contents, g (Kg/m^3) = 1000.00 (*Assumed as water density*)

Static Pressure Head = g x 9.8067 x H

= 1000 x 9.8067 x 1317/1000 = 12915.42 Pascal

= 12915.42×10^{-6} Mpa

= 0.012Mpa

Design Internal Pressure = Design Pressure + Pressure due to Static Head

= 0.1 MPa + 0.012 MPa

= 0.112 MPa

4.4 Hydrostatic Test Pressure [Ref: UG-99 (b)]

Hydrostatic Test Pressure = Maximum allowable working pressure x LSR x 1.3

LSR = Lowest Stress Ratio (From B.1)

LSR = Stress value at test Temperature / Stress value at design Temperature = 1

Hydrostatic Test Pressure for vacuum vessel = 0.10 x 1 x 1.3

= 0.13 MPa

Hydrotest to be carried out in Horizontal position only.

4.5 MATERIAL OF CONSTRUCTION EVALUATION CHART AS PER UG-23, UHA-23, TABLE UHA 23 OF ASME SEC VIII DIV-1 AND ASME Sec-II Part D,TABLE 1A

No	Component type	Material of construction	Allowable Stress (MPa)		
			MDMT (0^{O}C)	Design Temp (40 ^{O}C)	Hydrostatic temp (48^{O}C)
1	Inner & Outer Shell of vacuum chamber	SA 240 Type 304	138	138	138
2	Inner & Outer Shell Flanges of vacuum vessel	SA 240 Type 304L	115	115	115
3	Front bolted flange & Rear welded flange	SA 240 Type 304	138	138	138
4	Nozzle	SA 240 TP 304L	115	115	115
5	Nozzle Flanges	SA 240 Type 304L	115	115	115
6	Bolting for Vacuum vessel	SA 193 B8	172	172	172

4.6 Design of Outer cylindrical shell (Vacuum Vessel) Thickness under External Pressure [Ref: UG-28 (c) (1)]

The material selected for the outer cylindrical shell is SA 240 TYPE 304. Assumed thickness (t) of the vessel is 8 mm. Considering the mill tolerance of 0.25 mm from the table A 3.5, SEC II Part A, so effective thickness is 8 – 0.25 = 7.75 mm.

Outer diameter of the vessel (D_0) = 1200 mm

Inside diameter of the vessel (D) = 1184 mm

Length of the vessel = 1060 mm

The ratio of length of the vessel to the outer diameter of the vessel = L/Do = 1060/1200 = 0.88

The ratio of the outer diameter of the vessel to the thickness of the vessel = Do/t = 1200/8 = 150

From the ASME code Fig- g, Section II, Part D, Factor (A) = 0.00082

From the ASME code fig-HA-1, for SA 240 TYPE 304, Factor (B) = 58

The allowable external pressure working pressure (Pa) from the code is given by:

$$Pa = 4B / (3*(Do/t))$$

$$= 4 \times 58 / (3 \times (1200/7.75))$$

$$= 0.50 \text{ MPa}$$

The actual working pressure is 0.1 MPa which is lower than the allowable external pressure of 0.50 MPa so the provided thickness (7.75 mm) is adequate.

4.7 Design of Inner cylindrical shell (Vacuum Vessel) Thickness under Internal Pressure [Ref: UG-27]

The material selected for the outer cylindrical shell is SA 240 TYPE 304. Assumed thickness (t) of the vessel is 8 mm. Considering the mill tolerance of 0.25 mm from the UG 16 (c), table A 3.5, SEC II Part A, so effective thickness is 8 – 0.25 = 7.75 mm.

Internal design pressure (P) = 0.1 MPa

Maximum allowable stress from section II part D (S) = 138 MPa

The outer diameter of the vessel (D_0) = 316 mm

Inside diameter of the vessel (D) = 300 mm

The inside radius of the vessel (R) = 150 mm (taking corrosion allowances is zero)

Longitudinal joint efficiency from UW-12 = 0.7

The minimum required thickness (t) as per ASME Appindix -1 is given by :

$$t = PR/((SE)-(0.6P))$$

$$= 0.1 \times 150 / (138 \times 0.7 - 0.6 \times 0.1)$$

$$= 0.15 \text{ mm}$$

But according to ASME Code, UG-16 (b) the minimum thickness of the shell must be 1/16 inch excluding the corrosion allowances i.e 1.5 mm.

Since required thickness is less than the provided thickness, i.e. 1.5 < 7.75 mm, So provided thickness is adequate.

4.8 Design of Outer Cylindrical Shell (Vacuum Vessel) Thickness under Internal Pressure [Ref: UG-27]

The material selected for the outer cylindrical shell is SA 240 TYPE 304. Assumed thickness (t) of the vessel is 8 mm. Considering the mill tolerance of 0.25 mm from the UG 16 (c), table A 3.5, SEC II Part A, so effective thickness is 8 – 0.25 = 7.75 mm.

Internal design pressure (P) = 0.1 MPa

Maximum allowable stress from section II part D (S) = 138 MPa

The outer diameter of the vessel (D_0) = 1200 mm

The outer radius of the vessel (Ro) = 600 mm

Inside diameter of the vessel (D) = 1184 mm

The inside radius of the vessel (R) = 592 mm (taking corrosion allowances is zero)

Longitudinal joint efficiency from UW-12 = 0.7

The minimum required thickness (t) as per ASME Appendix -1 is given by:

$$t = PR/((SE)-(0.6P))$$

$$= 0.1x592 / (138x0.7 - 0.6x0.1)$$

$$= 0.61 \text{ mm}$$

But according to ASME Code, UG-16 (b) the minimum thickness of the shell must be 1/16 inch excluding the corrosion allowances i.e 1.5 mm.

Since required thickness is less than the provided thickness, i.e. 1.5 < 7.75 mm, So provided thickness is adequate

4.9 Design of Front Bolted Flange (Vacuum Vessel) Thickness [Ref: UG-34 (2) Unstayed Flat Heads & Covers]

The internal design pressure (P) = 0.1 MPa

Maximum allowable stress from section II part D (S) = 138 MPa

Diameter of the opening of the shell (d) = 1184 mm

Required thickness of the shell wall (Tr) = 0.61 mm

Nominal thickness of the shell (Ts) = 8.0 mm

Ratio of Tr/ Ts = 0.61/ 8 = 0.08

Factor (C) depending on the method of attachment = 0.30

Joint efficiency for full Radiography (E) = 0.7

The minimum thickness (t) required for the flat unstayed head [UG-34 (2) eqn-(2)] is given by :

t = d x sqrt (CP/SE)

= 1184 x sqrt(0.3 x 0.1/ 138 x 0.7)

= 21 mm

Thus the provided thickness is 22 mm.

4.10 Design of Rear Welded Flange (Vacuum Vessel) Thickness [Ref: UG-34 (2) Unstayed Flat Heads & Covers]

The internal design pressure (P) = 0.1 MPa

Maximum allowable stress from section II part D (S) = 138 MPa

Diameter of the opening of the shell (d) = 1184 mm

Required thickness of the shell wall (Tr) = 0.61 mm

Nominal thickness of the shell (Ts) = 8.0 mm

Ratio of Tr/ Ts = 0.61/ 8 = 0.08

Factor (C) depending on the method of attachment = 0.20

Joint efficiency for full Radiography (E) = 0.7

The minimum thickness (t) required for the flat unstayed head [UG-34 (2) eqn-(2)] is given by :

t = d x sqrt (CP/SE)

= 1184 x sqrt(0.3 x 0.1/ 138 x 0.7)

= 17 mm

Thus the provided thickness is 20 mm.

4.11 Design of Inner Flange (Vacuum vessel) Thickness [Ref: SEC-VIII DIV-1, NON-MANDATORYAPPENDIX-Y]

Type of Flange Selected (Category 3 Flange and Class-1 Assembly)	**(Loose Type Flanges Without Hub) (Fig-Y-5.1.1 (C)**		
Outside diameter of flange	A	410.0	mm
Inside diameter of flange	B	300.0	mm
Bolt circle diameter	C	343.0	mm
Thickness of the flange (assumed)	t	**36.0**	mm
Diameter of location of gasket load reaction	G	372.0	mm
Bolt hole diameter	D	13	mm
Internal design pressure	P	0.11	N/mm2
Hub length	h	0.0	mm
Hub thickness (small end)	g0	0.0	mm
Hub thickness (large end)	g1	0.0	mm
No. of bolts	n	20	mm
Nominal bolt diameter	db	12.0	mm
Bolt stress area for single M12 bolt	Absingle	72.3	mm2
Total Bolt area for M12 bolt = n x Absingle	Ab	1446.0	mm2
Bolt circle aspect ratio = n x D/3.141xC	AR	0.241	

Factors: [Ref: Appendix-Y- 3]

Bolt hole flexibility factor	rB	0.0056	rB = 1/n (4/√(1-AR^2)xtan¯1(√(1+AR/1-AR)-π-2AR)

Integral flange factor	ho	0.0	h0 = SQRT (Bxg0)
Factor K	K	1.3667	K = A/B
B1 = B for loose type flange	B1	300	
Shape Factor	β	1.0717	β = (C+B1)/2xB1
Shape Factor	a	1.255	a= (A+C)/2xB1

Bolt Loads and Flange Moments: [Ref: Appendix-Y-4]

Total hydrostatic end force	H	11949.5	N	$H=0.785G^2P$
Gasket load due to seating pressure	H_G	0		For self sealing O-ring
Total joint contact surface compression load	Hp	0		Hp = 2b x 3.14GmP
Hydrostatic end force on area inside of flange	H_D	7771.5	N	$H_D = 0.785B^2P$
Difference between H and H_D	H_T	4178.0	N	$H_T = H-H_D$
Radial distance from the bolt circle to point of intersection of hub and back of flange	R	21.5	mm	$h_D = (C-B)/2 - g1$ From Appndx-2
Radial distance from the bolt circle to the circle on which H_D acts	h_D	21.5	mm	$h_D = (C-B)/2$ From Table 2-6 Appndx-2
Radial distance from gasket load reaction to the bolt circle	h_G	-14.5	mm	$h_G = (C-G)/2$ From Table 2-6 Appndx-2
Radial distance from the bolt circle to the circle on which HT acts	h_T	3.5	mm	$h_T = (h_D+h_G)/2$ From Table 2-6 Appndx-2
Radial distance from the bolt circle to the outer edge of flange or spacer whichever is less	Hc	33.5	mm	hc = (A-C)/2
	F'	0.0		For category 3 class 1 assembly
	Js	0.24		$1/B1[(2xhxD)/\beta + h_c/a] + 3.141xr_B$
	Jp	0.17		$1/B1[(hxD)/\beta + h_c/a] + 3.141xr_B$
Component of moment due to H_D	M_D	167087.25	N-mm	$M_D = H_Dh_D$
Component of moment due to H_T	M_T	14622.85	N-mm	$M_T = H_Th_T$
Component of moment due to H_G	M_G	0.0	N-mm	$M_G = H_Gh_G$
Moment dur to H_D, H_T, H_G	M_P	181710.10	N-mm	$M_P = M_D + M_T + M_G$

Flange moment due to Flange-Hub Interaction	M_S	0.0		
Slope of Flange at Inside Diameter Times E	$E\theta_B$	1.2		
Contact force between flanges at h_C	Hc	5424.2	N	$H_C = (M_P + M_S) / Hc$
Bolt load at operating condition	**Wm1**	**17373.6**	N	$Wm1 = H + H_C + H_G$
Operating Bolt stress	Σb	12	N/mm2	σb = WM1/Ab
spacer thickness	Ts	0.0	mm	
Calculated strain length of bolt	L	78	mm	l = 2t + ts + (1/2db)
Design Pre stress in bolts	Si	11.84	N/mm2	

Calculations of Flange Stresses: [Ref: Appendix-Y- 6.1]

Radial flange stress at bolt circle	S_R	0.00	N/mm^2	
Longitudinal hub stress	S_H	0.00	N/mm^2	
Tangential flange stress at inside diameter	**S_T**	**0.14**	N/mm^2	$S_T = t \times E\theta_B / B1$
Allowable bolt stress at atmospheric temp.	Sa	130	N/mm^2	For B8 Grade SA 193 Table 3
Allowable bolt stress at design temp.	Sb	130	N/mm^2	
Allowable design stress for material of flange at design temperature	**Sf**	**115**	N/mm^2	

As per Appndix-Y-7 Tangential Flange Stress S_T not greater than Allowable Design Stress SF (115 MPa) and Operating bolt stress σb is not greater than Allowable bolt stress Sb.

Therefore the provided thickness of flange is adequate.

4.12 Design of Outer Flange (Vacuum vessel) Thickness [Ref: SEC-VIII DIV-1, NON- MANDATORYAPPENDIX-Y]

Type of Flange Selected (Category 3 Flange and Class-1 Assembly)	**(Loose Type Flanges Without Hub) (Fig-Y-5.1.1 (C)**		
Outside diameter of flange	A	1184.0	mm
Inside diameter of flange	B	1110.0	mm
Bolt circle diameter	C	1158.0	mm
Thickness of the flange (assumed)	t	**36.0**	mm
Diameter of location of gasket load reaction	G	1129.0	mm
Bolt hole diameter	D	13	mm
Internal design pressure	P	0.11	N/mm2
Hub length	h	0.0	mm

Hub thickness (small end)	g0	0.0	mm
Hub thickness (large end)	g1	0.0	mm
No. of bolts	n	48	mm
Nominal bolt diameter	db	12.0	mm
Bolt stress area for single M12 bolt	Absingle	72.3	mm2
Total Bolt area for M12 bolt = n x Absingle	Ab	3470.4	mm2
Bolt circle aspect ratio = n x D/3.141xC	AR	0.172	

Factors: [Ref: Appendix-Y- 3]

Bolt hole flexibility factor	rB	0.0081	rB = 1/n (4/√(1-AR^2)xtan¯1(√(1+AR/1-AR)-π-2AR)
Integral flange factor	ho	0.0	h0 = SQRT (Bxg0)
Factor K	K	1.0667	K = A/B
B1 = B for loose type flange	B1	1110.0	
Shape Factor	β	1.0216	β = (C+B1)/2xB1
Shape Factor	a	1.055	a= (A+C)/2xB1

Bolt Loads and Flange Moments: [Ref: Appendix-Y-4]

Total hydrostatic end force	H	110065.3	N	$H=0.785G^2P$
Gasket load due to seating pressure	H_G	0		For self sealing O-ring
Total joint contact surface compression load	Hp	0		Hp = 2b x 3.14GmP
Hydrostatic end force on area inside of flange	H_D	106391.8	N	$H_D = 0.785B^2P$
Difference between H and H_D	H_T	3673.4	N	$H_T = H-H_D$
Radial distance from the bolt circle to point of intersection of hub and back of flange	R	24.0	mm	h_D = (C-B)/2 - g1 From Appndx-2
Radial distance from the bolt circle to the circle on which H_D acts	h_D	24.0	mm	h_D = (C-B)/2 From Table 2-6 Appndx-2
Radial distance from gasket load reaction to the bolt circle	h_G	14.5	mm	h_G = (C-G)/2 From Table 2-6 Appndx-2
Radial distance from the bolt circle to the circle on which HT acts	h_T	319.3	mm	$h_T = (h_D+h_G)/2$ From Table 2-6 Appndx-2
Radial distance from the bolt circle to the outer edge of flange or spacer whichever is less	Hc	13.0	mm	hc = (A-C)/2

	F'	0.0		For category 3 class 1 assembly
	Js	0.08		$1/B1[(2xhxD)/\beta + h_c/a] + 3.141xr_B$
	Jp	0.06		$1/B1[(hxD)/\beta + h_c/a] + 3.141xr_B$
Component of moment due to H_D	M_D	2553404.04	N-mm	$M_D = H_D h_D$
Component of moment due to H_T	M_T	70713.25	N-mm	$M_T = H_T h_T$
Component of moment due to H_G	M_G	0.0	N-mm	$M_G = H_G h_G$
Moment dur to H_D, H_T, H_G	M_P	2624117.29	N-mm	$M_P = M_D + M_T + M_G$
Flange moment due to Flange-Hub Interaction	M_S	0.0		
Slope of Flange at Inside Diameter Times E	$E\theta_B$	5.6		
Contact force between flanges at h_C	Hc	201855.2	N	$H_C = (M_P + M_S) / Hc$
Bolt load at operating condition	Wm1	311920.4	N	$Wm1 = H + H_C + H_G$
Operating Bolt stress	Σb	89.9	N/mm2	σb = WM1/Ab
spacer thickness	Ts	0.0	mm	
Calculated strain length of bolt	L	78.0	mm	l = 2t + ts + (1/2db)
Design Pre stress in bolts	Si	89.76	N/mm2	

Calculations of Flange Stresses: [Ref: Appendix-Y- 6.1]

Radial flange stress at bolt circle	S_R	0.00	N/mm^2	
Longitudinal hub stress	S_H	0.00	N/mm^2	
Tangential flange stress at inside diameter	**S_T**	**0.18**	N/mm^2	$S_T = t \times E\theta_B / B1$
Allowable bolt stress at atmospheric temp.	Sa	130	N/mm^2	For B8 Grade SA 193 Table 3
Allowable bolt stress at design temp.	Sb	130	N/mm^2	
Allowable design stress for material of flange at design temperature	**Sf**	**115**	N/mm^2	

As per Appndix-Y-7 Tangential Flange Stress S_T not greater than Allowable Design Stress Sf (115 MPa) and Operating bolt stress σb is not greater than Allowable bolt stress Sb.

Therefore the provided thickness of flange is adequate.

Bolt Spacing Requirement: (Appendix 2-5 (d))

Nominal bolt diameter	A	12	mm	For M12 bolt
Actual Thickness of flange	T	40	mm	
Gasket factor	M	0		For o-rings, m=0 (From Table 2-5.1)
Maximum bolt spacing from the equation (3) = 2x12 + (6x40/(0+0.5)) = 480	Bsmax	504	mm	Bsmax = 2a+(6t/(m+0.5))
Actual spacing between adjacent bolts (provided)	Bs	76	mm	
Therefore bolt spacing provided is adequate				

4.13 Openings in Vacuum Vessel [Ref: UG-36]

Nozzle	Location	Size (NPS)	OD (mm)	Schedule	Thk	Effective ID
N1	Shell	8	219.1	40	8.18	202.74
N2	Shell	5	141.3	40	6.55	128.2

4.14 Thickness of Nozzle (N1) Wall under Internal Pressure [Appendix 1-1(a)]

Pipe selected: NPS 8, SCH 40, OD=219.1, 8.18 Thick

The material selected for the outer cylindrical nozzle is SA 240 TYPE 304L. Assumed thickness (t) of the nozzle is 8.18 mm Considering Mill Under tolerance of 12.5 % (Table-3, Sec-II, Part A), so effective thickness is = 8.18 x 0.875 = 7.15 mm.

Internal design pressure (P) = 0.1 MPa

Maximum allowable stress from section II part D (S) = 115 MPa

Inside diameter of the nozzle (D) = 202.74 mm

The inside radius of the nozzle (R) = 101.37 mm (taking corrosion allowances is zero)

Longitudinal joint efficiency from UW-12 = 1

The minimum required thickness (t) as per ASME Appendix -1 is given by :

$$t = PR/((SE)-(0.4P))$$

$$= 0.1x101.37/ (115x0.7 + 0.4x0.1)$$

$$= 0.126 \text{ mm}$$

But according to ASME Code, UG-16 (b) the minimum thickness of the nozzle must exclude the corrosion allowances is 1.5 mm.

Since the required thickness is less than the provided thickness, i.e. 1.5 < 7.15 mm, So provided thickness is adequate.

4.15 Thickness of Nozzle (N2) Wall under Internal Pressure [Appendix 1-1(a)]

Pipe selected: NPS 5 SCH 40, OD=141.3, 6.55 Thick

The material selected for the outer cylindrical nozzle is SA 240 TYPE 304L. Assumed thickness (t) of the nozzle is 6.55 mm Considering Mill Under tolerance of 12.5 % (Table-3, Sec-II, Part A), so effective thickness is = 6.55 x 0.875 = 5.73 mm.

Internal design pressure (P) = 0.1 MPa

Maximum allowable stress from section II part D (S) = 115 MPa

Inside diameter of the nozzle (D) = 128.2 mm

The inside radius of the nozzle (R) = 114.1 mm (taking corrosion allowances is zero)

Longitudinal joint efficiency from UW-12 = 1

The minimum required thickness (t) as per ASME Appendix -1 is given by :

$$t = PR/((SE)-(0.4P))$$

$$= 0.1 \times 64.1/ (115x0.7 + 0.4x0.1)$$

$$= 0.07 \text{ mm}$$

But according to ASME Code, UG-16 (b) the minimum thickness of the nozzle must excludi the corrosion allowances is 1.5 mm.

Since required thickness is less than the provided thickness, i.e. 1.5 < 5.73 mm, So provided thickness is adequate.

4.16 Stress analysis of lifting lug for the vacuum vessel chamber

Material of the lifting lug is SA 240 TYPE 304

Allowable stress for the lifting lug (S) = 138 MPa

Thickness of the lug (t) = 15.0 mm

Hole diameter of the lug (d) = 20 mm

Outer radious of the lug (r) = 30 mm

Hole center height from the base of the lug (H) = 40 mm

Weld lug size = 6 mm

Empty weight of the container = 20000 N

Number of the lug (n) = 2

Load per lug = 10000 N

Considering the impact factor of 1.5, then total load per lug = 10000 x 1.5 = 15000 N

The area resisting the tensile load (A1) =2x(t x (r-d/2))

= 2 x (15 x (30-20/2))

= 600 mm^2

Tensile stress on the lifting lug ($T_{tensile}$) = 15000/ 600 = 25 MPa

Pin bearing area (A2) = π x d x t

= 3.14 x 20 x 15 = 942.3 mm^2

Bearing stress on the pin = 15000/ 942.3

= 15.915 MPa

Area resisting to shear (A3) = w x t = 100 x 15 = 1500 mm^2

Shear stress on the lug = 15000/ 1500 =10 MPa

CHAPTER 5

5.0 Thermal Design of Nitrogen Storage Container

Proper thermal designing in cryogenic systems is the most challenging design for cryogenic apparatus like liquid nitrogen storage containers, which directly impacts on the performance during its operation. The thermal design should be such that heat loss will be minimized as much as possible and it should be in the acceptable range. The heat transfer basically occurs in three ways;

Conduction - heat transfer through solids

Convection - heat transfer through liquids and gases

Radiation – heat transfer through space

Heat transfer through conduction can be more precisely calculated, but in the case heat transfer through convection and radiation a reasonable approximate estimate can be done.

In liquid Helium or Nitrogen storage container, it needs to have the least amount of heat load coming in to that storage container (at 4.2 K or 77K). The maximum contribution of heat load that can be transferred in to the storage container is due to natural convection from the atmospheric air which is at 300 K, hence to reduce the heat load due to natural convection we need to evacuate the space between vacuum chamber and the nitrogen storage chamber. The space can be evacuated using roughing pump and the turbo pump to create a vacuum in the range of 10^{-5} mbar. The number of gas molecules reduces at vacuum condition which increases the mean free path of the gas molecules, approximately at 10^-4 mbar , mean free path is about 100 cm which is more than the distance between the surfaces which leads to reduce heat load to few milliwatts at vacuum of 10^{-5} mbar. Multilayer insulation, which consists of highly reflective aluminum sheets are intended to reduce radiation heat transfer. Insulator like G10 material is used as the separators between the nitrogen chamber and the vacuum chamber for reducing the conduction heat load.

In nitrogen container vessel, total heat loss is mainly due to the conduction, radiation only.

Heat loss due to the natural convection is negligible due to evacuate the vacuum chamber. The conduction heat transfer occurred due to G10 separators and due to neck region of cylindrical surface. Similarly radiation heat transferred due to both flat end of the vacuum chamber, due to cylindrical surface of the vacuum chamber, due to flat end of the neck and due to cylindrical surface of the neck.

5.1 Conduction heat load calculation

The conduction heat load which can transfer to the nitrogen container through the neck portion of the cylinder surfaces. One dimensional heat transfer equation for conduction,

$$\dot{Q} = -K.A.dT/dX \qquad (1)$$

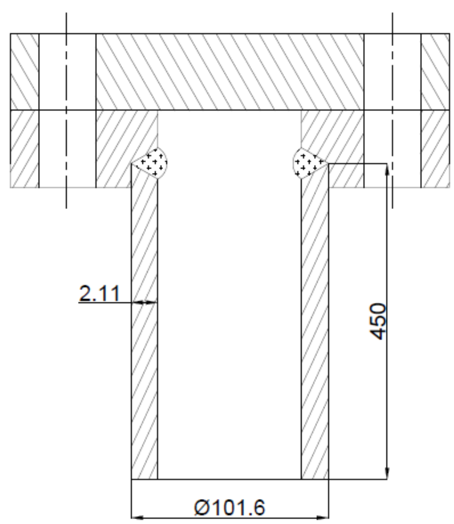


Fig. 5.1 Neck of the nitrogen chamber

Here Q is heat transferred due to conduction A is area of cross-section perpendicular to the temperature flow line. Since the thermal conductivity of Stainless steel varies with temperature, thus it is better to take the value of mean thermal conductivity from low temperature to higher temperature.

$$\overline{K} = \int_{T1}^{T2} TdT \qquad (2)$$

With the mean thermal conductivity considered the heat transfer equation becomes,

$$\dot{Q} = \overline{K} \text{ x } \frac{A}{L} \qquad (3)$$

Conduction heat load on the nitrogen chamber due to following reasons:

- Due to G10 separators
- Due to neck region of cylindrical surface

5.1.1 Conduction heat load due to G10 supports:

Outer diameter of G10 separator = 40 mm = 0.04 m

Inner Diameter of G10 separator = 25 mm = 0.025 m

A = Area exposed for conduction = $\frac{\pi}{4}$x$(0.04^2 - 0.025^2)$= 0.000765 m2

dx = Thickness of the G10 separator = 110 mm = 0.011 m

K = thermal conductivity of G10 = 0.5 W.$m^{-1}k^{-1}$

Atmospheric temperature T1 = 300K

Liquid nitrogen temp. T2 = 77 K

Total number of separator = 15

The conduction heat transfer due to G10 support Qc1 = K.A.$\frac{dT}{dx}$

Qc1 = 0.5 x 0.000765 x 15 x $\frac{(300-77)}{0.110}$

Qc1 = 11.6 watt

5.1.2 Conduction heat load due to neck region of cylindrical surface:

We have a SS304 neck with ID= 97.38 mm and OD = 101.6 mm and length 450 mm. one end is at 300K and the other at 80K for the maximum possible heat load that can be transferred to the liquid nitrogen.

For SS304 mean thermal conductivity from 300 to 80 K = 2711W/m (from Experimental Temperature for Low-temperature measurements by Jack.W.Ekin)

Cross-sectional area of the pipe A = $\frac{\pi}{4}$x$(0.101^2 - 0.09738^2)$= $0.563*10^{-3}$ m2

Therefore conduction heat transferred through the neck Qc2 = $\overline{K}.\frac{A}{L}$

Qc2 = 2711x$\frac{0.563*10^{-3}}{0.45}$

Qc2 = 3.39 W

Total heat loss due to conduction Qc = Qc1 + Qc2

= 14.99 watt

5.2 Radiation heat load calculation

Radiation heat transfer takes place in space in the form of electromagnetic waves.

When a hot body radiating energy to the surrounding cooler body, then the net heat transferred according to ***Stefan-Boltzmann Law*** occurred due to radiation is given below.

$q = \varepsilon\ \sigma\ (T_h^4 - T_c^4)\ A$

where,

q = heat transfer per unit time due to radiation

T_h = hot body temperature

T_c = cold surroundings temperature

A = area of the object exposed to radiation (m^2)

$\sigma = 5.6703\ 10^{-8}\ (W/m^2K^4)$ - ***The Stefan-Boltzmann Constant***

T = absolute temperature Kelvin (K)

5.2.1 Radiation heat load due to flat end of vacuum chamber:

D = Diameter of the flat end exposed to radiation = 1100 mm = 1.1 m

Atmospheric temperature T1 = 300K

Liquid nitrogen temp. T2 = 77 K

A = Area exposed to radiation = $\frac{\pi}{4}$x1.1^2 = 0.95 m2

Emissivity of multilayer insulation aluminum foil = 0.05

No of layers of insulation = 46

Radiation heat transfer Q_{R1} = σ x A x ϵ x ($T1^4$ - $T2^4$), where σ = 5.67x 10^{-8} W m^{-2} K^{-4}

Q_{R1} = 5.67x 10^{-8} x 0.95 x $\frac{0.05}{46}$ x (300^4 -77^4)

= 0.47 Watt

5.2.2 Radiation heat load due to cylindrical surface of vacuum chamber:

D = Outer diameter of the vacuum chamber = 0.1016 m

L = Length of the neck = 1 m

Outer surface area exposed to radiation (A) = π x D x L

A = 3.45 m^2

Atmospheric temperature T1 = 300K

Liquid nitrogen temp. T2 = 77 K

Emissivity of multilayer insulation aluminum foil = 0.05

No of layers of insulation = 46

Radiation heat transfer Q_{R2} = σ x A x ϵ x ($T1^4$ - $T2^4$), where σ = 5.67x 10^{-8} W m^{-2} K^{-4}

Q_{R2} = 5.67x 10^{-8} x 3.45 x $\frac{0.05}{46}$ x (300^4 -77^4)

Q_{R2} = 0.95 watt

5.2.3 Radiation heat load due to flat end of neck portion:

D = Diameter of the flat end exposed to radiation = 101.6mm = 0.1016m

Atmospheric temperature T1 = 3000K

Liquid nitrogen temp. T2 = 77 K

Area A = $\frac{\pi}{4}$x0.101^2 = 0.0081 m2

Radiation heat transfer Q_{R3} = σ x A x ϵ x ($T1^4$ - $T2^4$), where σ = 5.67x 10^{-8} W m^{-2} K^{-4}

Q_{R3} = 5.67x 10^{-8} x 0.0081 x 0.02 x (300^4 -77^4)

= 0.074 Watt

5.2.4 Radiation heat load due to cylindrical surface of neck portion:

D = Outer diameter of the neck = 0.1016 m

L = Length of the neck = 0.45 m

Outer surface area exposed to radiation (A) = π x D x L

A = 0.143 m^2

Atmospheric temperature T1 = 300K

Liquid nitrogen temp. T2 = 77 K

Emissivity of multilayer insulation aluminum foil = 0.05

No of layers of insulation = 30

Radiation heat transfer Q_{R4} = σ x A x ϵ x ($T1^4$ - $T2^4$), where σ = 5.67x 10^{-8} W m^{-2} K^{-4}

Q_{R4} = 5.67x 10^{-8} x 0.143 x $\frac{0.05}{30}$ x (300^4 -77^4)

Q_{R4} = 0.1 watt

Total heat loss due to radiation Q_R = Q_{R1} + Q_{R2} + Q_{R3} + Q_{R4}

= 0.47 + 0.95 +0.074 + 0.1

= 1.59 watt

Total Heat load Q = Qc + Q_R

= 14.99 + 1.59

= 16.58 watts

Total heat load Q = $\dot{m}.h_f$

Where h_f is latent heat of vaporization and $\dot{m}$ is mass flow rate

Latent heat of vaporization of liquid nitrogen = 199 J/s

1 liter of liquid nitrogen = 0.8 kg

I kg of liquid nitrogen = 1.25 liters

Mass flow rate ($\dot{m}$) = 16.58/199 = 0.08 g/s =0.28 kg/hr = 0.35 liters/ hr

Thus the rate of evaporation of liquid nitrogen is about 0.35 liters per hour.

CHAPTER 6

6.1 **CAD and Finite Element Method**

After finalizing dimensions like the thickness of individual components from ASME design code, 2D concept layout drawing (as sown in fig. 6.1) has been developed in AUTOCAD software and consequently we developed the 3D model (as shown in fig. 6.2) in PRO-E software which was exported to ANSYS for Finite Element Analysis purposes. Solid meshing was done on the full assembly and proper contact parameters were defined between the individual components. Separate material properties were assigned for different types of materials. Internal pressure has been applied to the inner nitrogen chamber and external pressure applied to the outer vacuum chamber.

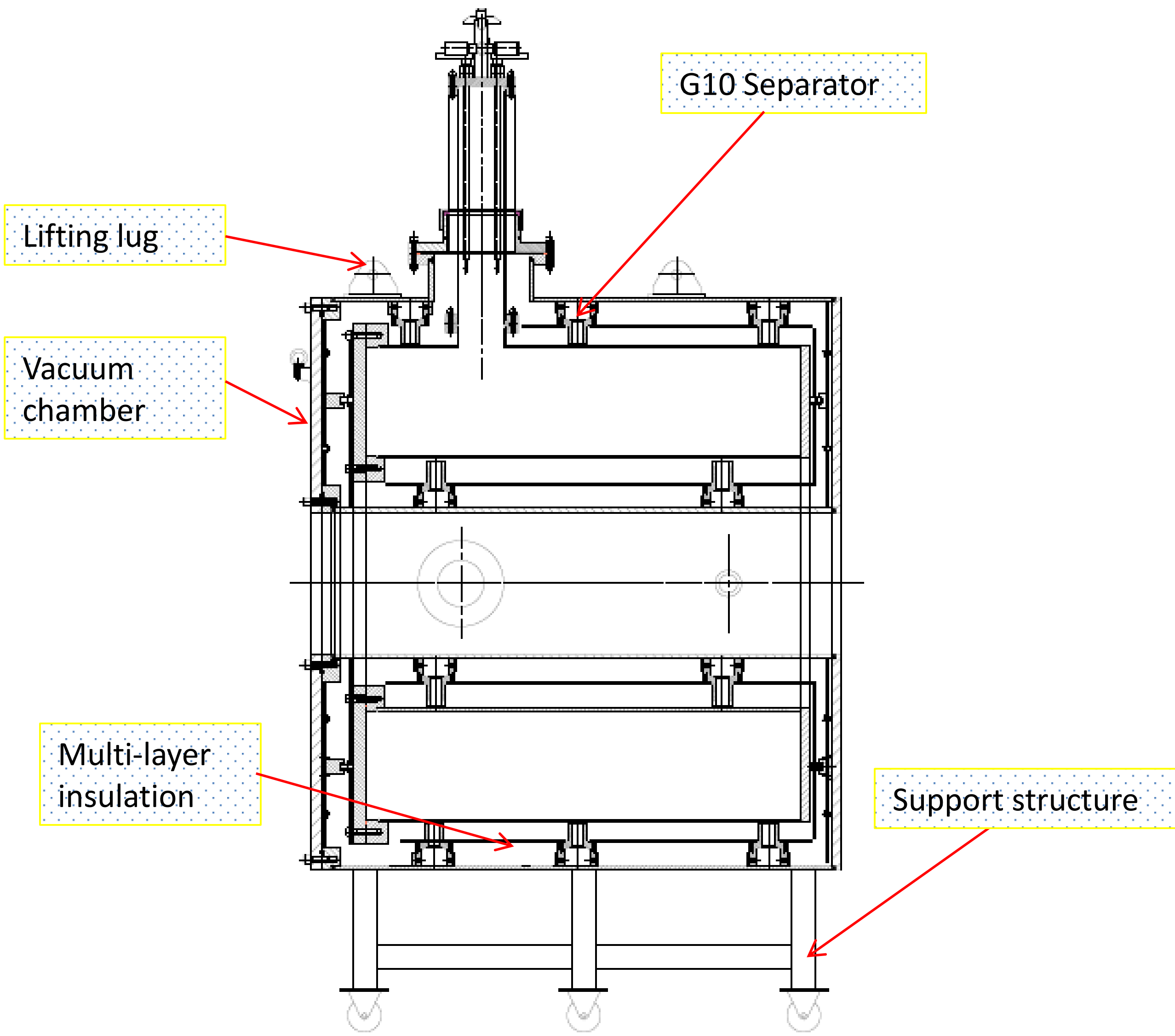
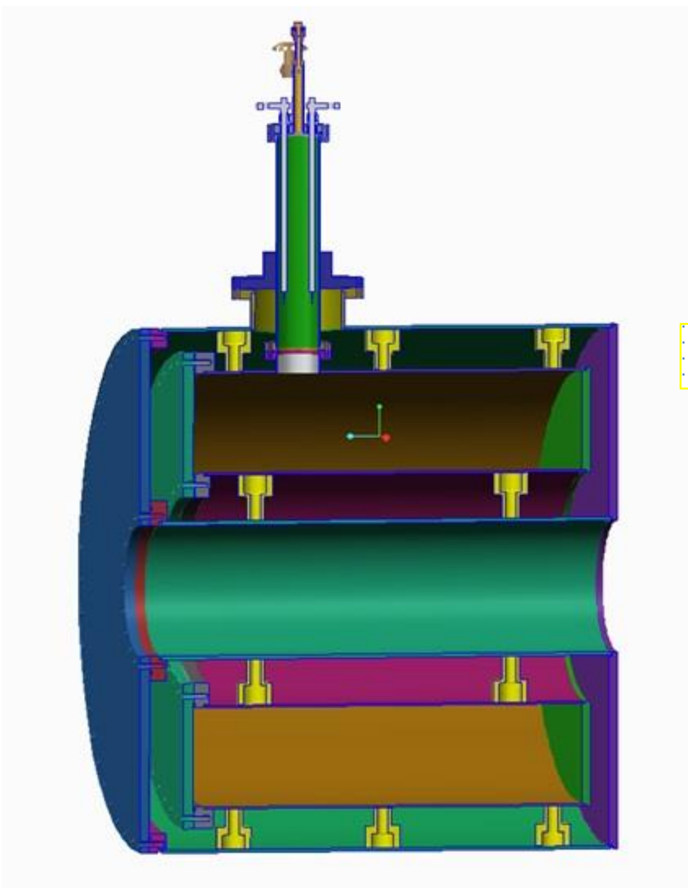


Fig.6.1: 2D Layout concept drawing

In layout drawing twenty numbers of G10 separators are used in between the nitrogen chamber and the vacuum chamber for the reduce conduction heat transfer. Some multilayer insulation having high reflectivity and low thermal conductivity are used to reduce radiation heat transfer. Multilayer insulation consists of the multiple layer of thin aluminum sheets of thickness 5 – 12 nm on Mylar film. Rear end covers are connected with the both nitrogen vessel and vacuum vessel through the welding process, whereas front end covers are attached with the vessels through the bolt joints. Front flange rings are also connected with vessel through weld joints.



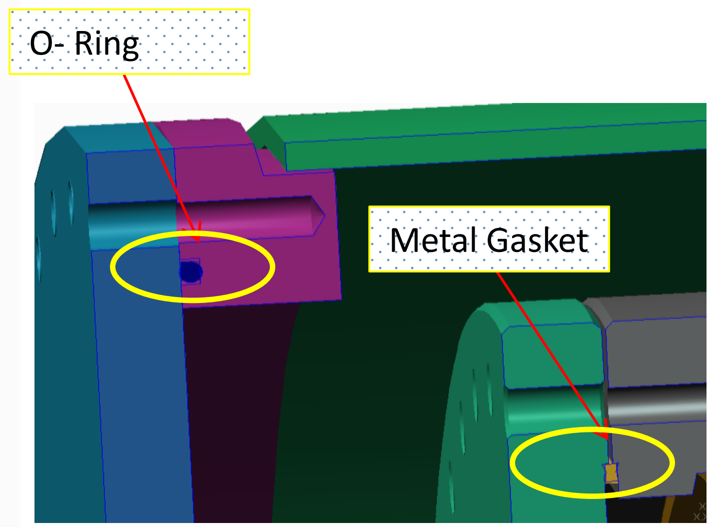


Fig. 6.2 cross section of main assembly

Fig. 6.3 Sealing arrangement in assembly

Fig 6.3 shows the leak proof joint arrangement for both the nitrogen storage vessel and vacuum vessel. In nitrogen storage vessel metal gasket made up from copper material used and for vacuum vessel O-ring made up from the Viton. Metal gaskets are used in cryogenic reservoir rather than O-ring because O-ring may get permanent deformation at low temperature, which leads to leakage in vessel and heat loss.

6.1: Finite Element Analysis Plots

Finite element analysis was performed by ANSYS 11.0 software; first we imported the 3D model from the Pro-E software. All the fasteners, Sharpe edges and small surfaces are removed from the main model for achieving a fine mesh model. Proper material properties like young modulus and Poisson's ratios for different types of materials have been assigned separately. Proper contact parameters are also assigned for different joints. Solid92 element is used for model meshing. 1 bar external pressure has applied on the vacuum chamber and 2 bar internal pressure applied on the nitrogen vessel and the portion surfaces which are resting over the support assembly are fixed as shown in fig 6.4. it is observed that the maximum 102 Mpa stress (fig. 6.6) developed on the vessel assembly which is well below the yield stress of the specified material. Total displacement (Fig. 6.5) on the vessel during loading is 0.14 mm which is very small.

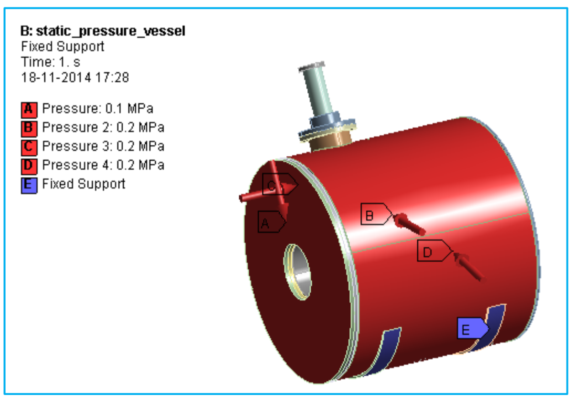


Fig. 6.4 Loading and Boundary condition

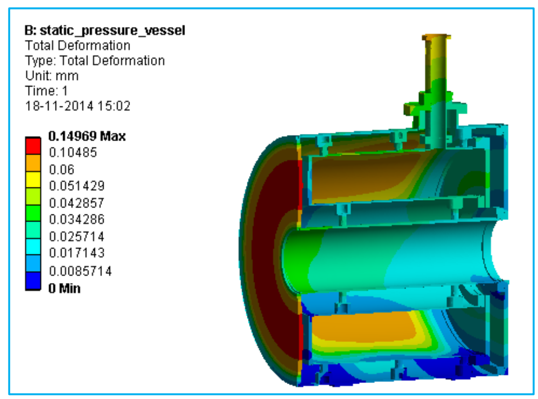


Fig. 6.5 Total displacement plot

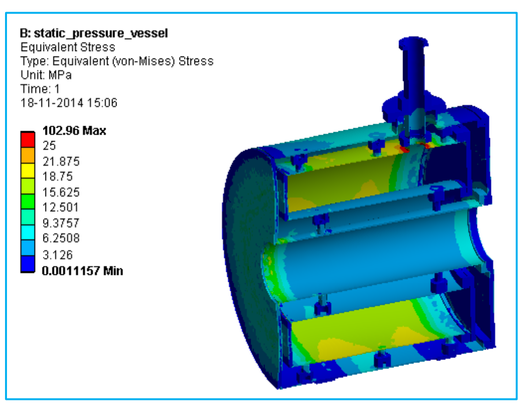


Fig 6.6 Von mises stress plot

6.2: Nitrogen chamber weld joint details

Nitrogen chamber has inner and outer shells which are connected with end cover plates, nozzle and pipes through the welding. ASME code section IX has been used for weld details. Fig 6.2 shows the different types of weld joints used in fabrication of nitrogen chamber.

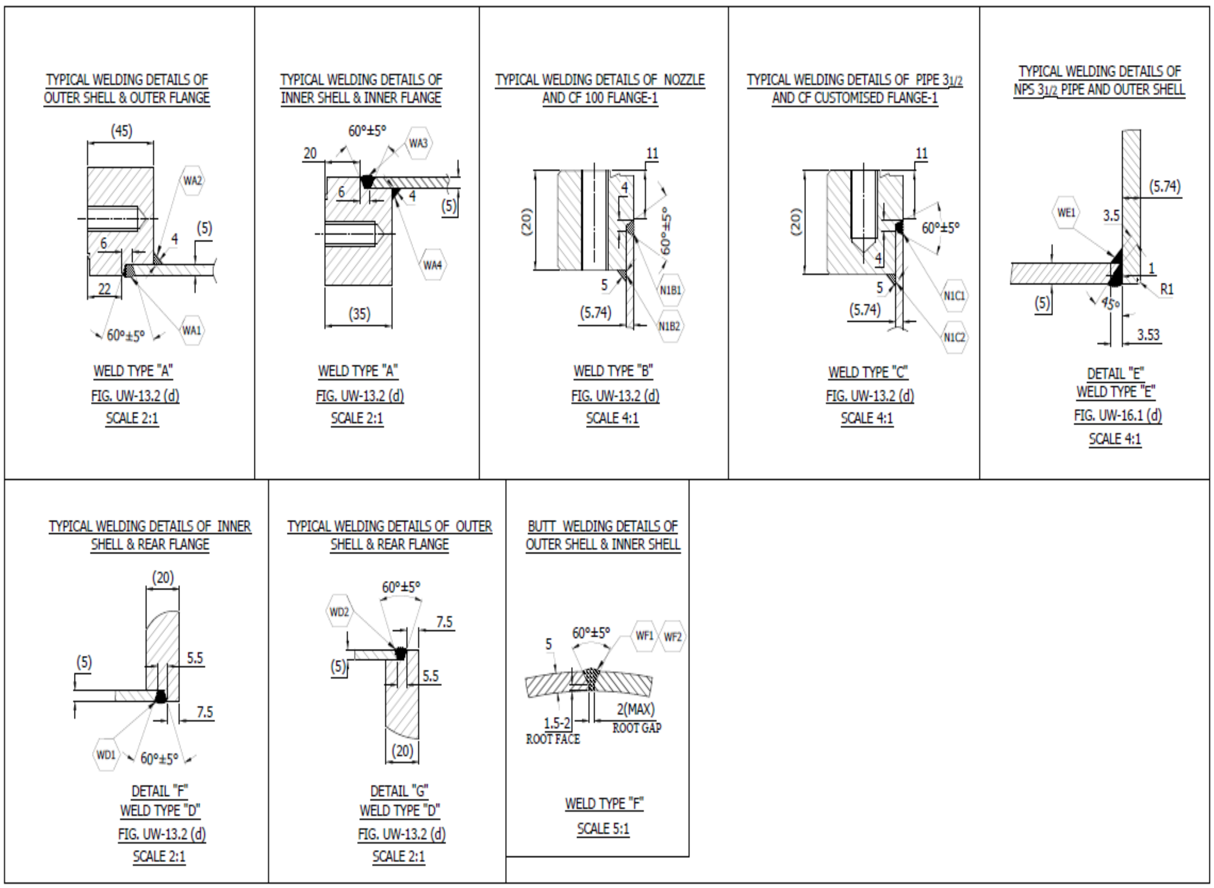


Fig 6.7 weld joints details in nitrogen chamber[1]

6.3: Vacuum chamber welds joint details

Vacuum chamber has inner and outer shells which are connected with end cover plates, nozzle and pipes through the welding. ASME code section IX has been used for weld details. Fig 6.2 shows the different types of weld joints used in fabrication of vacuum chamber.

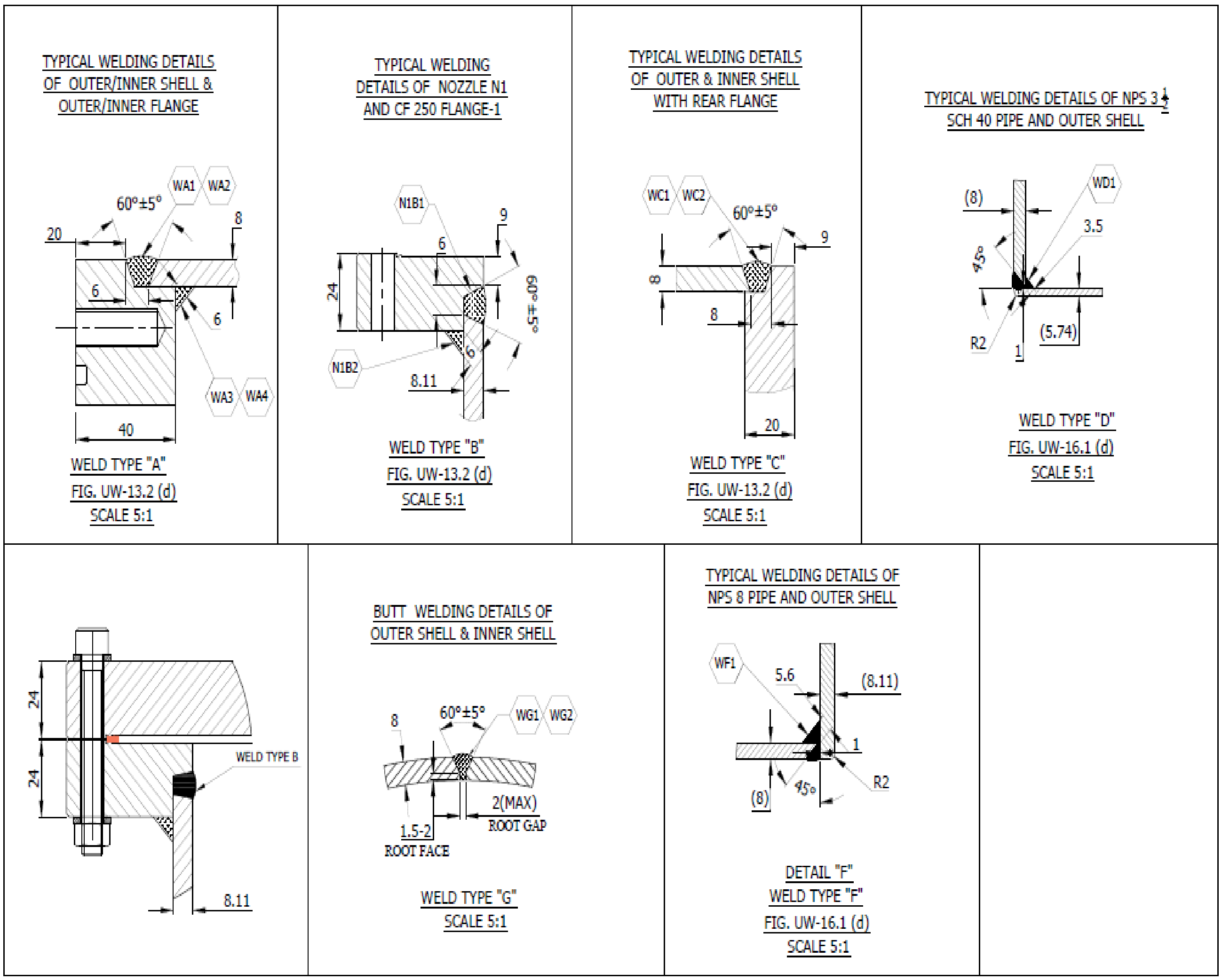


Fig 6.8 weld joints details in vacuum chamber [1]

CHAPTER 7

7. Fabrication and Assembly of Vessel

All the components are manufactured as per the part drawing. Dimensional tolerances and surface finish are the critical parameter during the assembly of the components. During assembly front cover end is connected with vessel through the bolt joints. Nozzles are connected with the vessel through the TIG welding and CF flanges are connected with the nozzle through the TIG welding. The following fig. 7.1 shows the liquid nitrogen vessel during assembly.

Fig. 7.1 Assembly of LN2 Storage Container

7.1 Experimental Test

Hydrostatic test and leak proof test are the two important tests which are to be carried out. Hydrostatic test is required for ensuring the strength of the vessel during actual loading also to detect leakage. Hydrostatic test is done on the nitrogen vessel assembly as well as vacuum vessel assembly separately. For hydrostatic test, Nitrogen chamber assembly was pressurized at 2.6 bar for 30 minutes and vacuum vessel assembly was pressurized at 1.3 bar for 30 minutes. For ensuring the effectiveness of leak proof joints, leak test was carried out. Roughing pump and turbo pumps are used for achieving a vacuum of order 10^{-5} in a vacuum vessel as shown in the fig. 7.2 for minimum evaporation of liquid nitrogen.

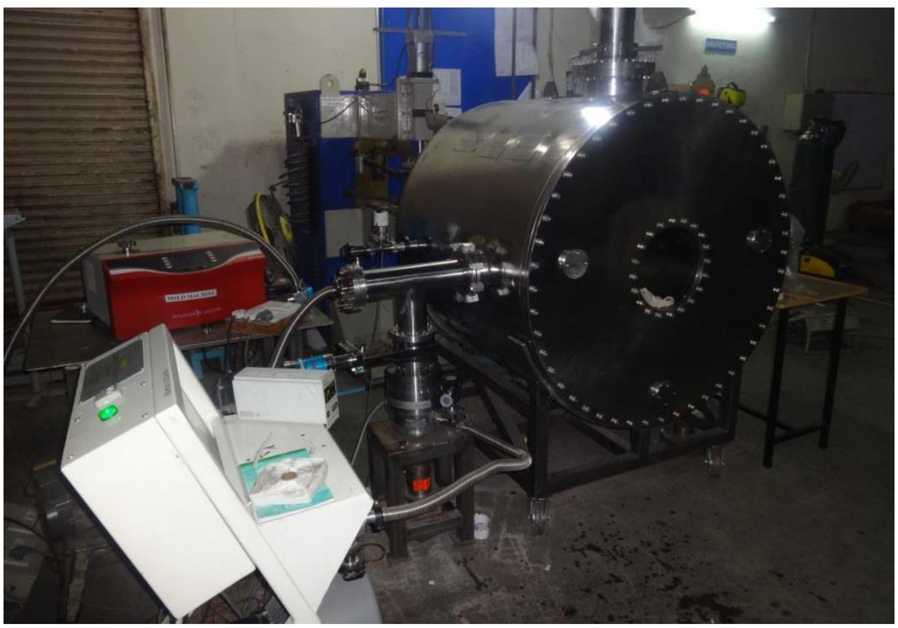


Fig. 7.2: Vacuum leak test setup

Fig. 7.3: Vacuum pressure gauge

CHAPTER 8

8.1 Discussion and Conclusions

- The liquid nitrogen storage vessel has been designed as per ASME Boiler and Pressure Vessel Code. ASME section II used for material selection, section V used for nondestructive testing like weld defect detection, section VIII division 1 used for design of components and section IX used for welding and brazing qualification.
- Finite Element Analysis of the assembly was carried out by ANSYS workbench and found that maximum Vonmises stress was 102 MPa which is well below the yield limit of the specified material, thus the design is safe.
- Total heat load transferred to the nitrogen chamber is minimized by implementing multilayer insulation for the reduction of radiation heat transfer, G10 insulator separator for reduction in conduction heat transfer and a evacuated chamber of order 10^{-5} mbar for the reduction in convection heat transfer. The total heat load on the nitrogen chamber is 16.58 watts, which shows the effective thermal design of the nitrogen storage container.
- Hydrostatic test was carried out through the water medium on both nitrogen chamber assembly and the vacuum vessel assembly separately for ensuring the strength against the specified load and it is found that both the assembly passed the required criteria.
- To find any defect in welding and material, cold shock test was carryout five times on the assembly and it is observed that there was no crack or defect found on the material as well as in welding.
- For minimum evaporation of the liquid nitrogen, a higher order (10^{-5}) of vacuum was created by the vacuum chamber over the nitrogen chamber, metal gasket (Cu) used in nitrogen chamber and O-ring (NBR) used in vacuum chamber was used for leak proof joints.

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